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CONTRIBUTION OF DOWNSTREAM STATOR TO THE INTERACTION NOISE OF A SINGLE-STAGE AXIAL-FLOW COMPRESSOR

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SUMMARY

Some experimental results are presented of a noise study of a single-stage axial-flow compressor tested in an anechoic chamber. The objective of the study was to investigate the effect of the contribution of the downstream stator on the interaction noise of the compressor. A variable-speed electric motor was used to drive the compressor rotor through a range of rotor tip Mach numbers from 0.20 to 0.50. Provisions were made for testing separately the inlet-guide-vane—rotor combination, the rotor-stator combination, and the inlet-guide-vane—rotor—stator combination. The test setup included capability for increasing the spacing between the inlet guide vanes and the rotor.

The data are presented in the form of noise radiation patterns and frequency spectra. These data show that the noise levels due to the interactions of rotor and downstream stator are appreciably less than those due to inlet guide vane (IGV) and rotor interactions. Noise reductions result from increased IGV—rotor spacing for the IGV—rotor configuration, and these reductions are further increased with the addition of the downstream stator. It has been possible to relate variations in the noise levels with a knowledge of local flow conditions in the vicinity of the blades.

INTRODUCTION

Studies of noise generation in axial-flow compressors (refs. 1, 2, and 3) have shown that the predominant sources of discrete frequency noise are the interactions between the velocity fields of the fixed and moving surfaces. The experimental data of reference 1, obtained on a single-stage axial-flow compressor, showed an appreciable reduction in the noise level of the discrete tone due to increased spacing between the inlet guide vane (IGV) and rotor. A theoretical treatment of this interaction noise (ref. 3) indicates that the physical characteristics of the noise generation mechanism for the IGV—rotor combination are the same as those for the rotor-stator combination.

Information in this paper is concerned with the extension of the experimental work of reference 1 and includes the contribution of the downstream stator on the interaction noise of the rotor-stator combination as well as the IGV—rotor—stator combination. The relative importance of spacing between the IGV and rotor and between the rotor and stator as a means of reducing the compressor noise is discussed.

SYMBOLS

c	blade chord, inches (centimeters)
d	projected blade width (in axial direction)
f	frequency, cycles per second (1 cycle per second = 1 hertz)
K	Strouhal number
u	rotational tip velocity of rotor, feet per second (meters per second)
V_a	axial velocity at rotor blades, feet per second (meters per second)
V_l	local velocity in the wake, feet per second (meters per second)
V_r	relative velocity at rotor blade, feet per second (meters per second)
V_s	absolute velocity at downstream stator, feet per second (meters per second)
V_{tan}	tangential velocity at downstream stator, feet per second (meters per second)
V_∞	stream velocity, feet per second (meters per second)
x	distance from trailing edge of guide vane to leading edge of rotor blade, inches (centimeters)

APPARATUS AND METHODS

Description of Test Setup

The test setup for the present studies made use of the single-stage axial-flow compressor of reference 1 with the addition of a downstream stator. A photograph of the single-stage test compressor located in an anechoic test cell is shown in figure 1 along

with an insert sketch of the compressor and drive motor. The compressor was driven by a variable-speed electric motor at speeds up to a maximum of 8250 rpm, which corresponds to a rotor tip Mach number of 0.475.

The test compressor consisted of a single-stage axial-flow rotor having 53 blades with a tip diameter of 14.75 in. (37.5 cm), a hub-to-tip radius ratio of 0.727, two sets of interchangeable upstream vanes (inlet guide vane or IGV) having 53 and 62 vanes, respectively, and one set of downstream vanes (stator) consisting of 62 vanes. IGV, stator vanes, and rotor blades, which were taken from a multiple-stage compressor of a jet engine, had NACA 65-series airfoil sections with a constant chord of 0.70 in. (1.8 cm).

For the basic investigation, the stator assembly was installed downstream of the rotor at the same axial separation distance as the inlet guide vanes were upstream (0.535 chord). Provisions were made for separately testing the IGV—rotor combination, rotor-stator combination, and the IGV—rotor—stator combination. The test setup included capability for increasing the axial spacing between the IGV and rotor up to a maximum of 9.11 chords by use of spacer rings. The spacing between the rotor and stator was not varied.

Noise and Pressure Measurements

Noise measurements were obtained along a 12-ft-radius (3.66 m) circle every 15° of azimuth from 0° (ahead of the inlet) to 90°. In addition continuous noise recordings were made with a slowly moving microphone that traversed the azimuth range. Noise measuring and recording instrumentation, as well as measurement, calibration, and data reduction procedures, were similar to those described for the related studies of reference 1.

Compressor Operation

The compressor was operated under various test conditions to investigate the relative contributions of IGV and stators to the interaction noise that is radiated out the inlet duct. For each test configuration, noise data, along with measured airflow through the compressor, were obtained for three rotor speeds for purposes of comparison.

IGV and stator vanes were set for zero turning angle so that there were only small differences in the mean flow conditions at the rotor; that is, the rotor operated at approximately the same turning angle and at a given rotational speed absorbed nearly constant power for all configurations.

Figure 2 is a plot of weight flow in lbm/sec (kg/sec) as a function of the compressor speed for each of the configurations tested. The maximum weight flows for a given compressor speed were obtained with the rotor alone, as represented by the upper curve in figure 2, and the minimum weight flows were obtained for the IGV—rotor—stator

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combination, as represented by the lower curve in the figure. The weight flows for both the IGV—rotor and the rotor-stator configurations fell about half way between the above two extremes. This decrease in weight flow with the IGV and/or stator in place is due to duct losses.

RESULTS AND DISCUSSION

Noise Spectra

Figure 3 presents a one-third octave band spectrum of sound pressure level measured at 30° azimuth for the 62-IGV—rotor—stator combination operated at a rotor speed of 8250 rpm. Maximum noise band level is shown to be close to the overall sound pressure level value (shown at left of figure). This maximum level occurs in the frequency band corresponding to the rotor blade passage frequency of 7290 cps. Further analysis of this noise in the frequency range from 4000 to 8000 cps is shown in figure 4 as a narrow band spectrum obtained with a 50-cps constant bandwidth filter. The narrow band analysis of figure 4 emphasizes the predominance of discrete frequency noise associated with the fundamental blade passage frequency and also indicates a broadband peak of lower amplitude at about 5200 cps. This broadband peak occurring at the lower frequency and which was evident in the measured data for all configurations tested is believed to be due to vortex shedding from the rotor blades. Its frequency can be predicted by the following expression from reference 4:

$$f = K \frac{u}{d}$$

where the Strouhal number K equals 0.2 in this calculation, u is the rotor tip velocity, and d is the projected rotor blade width in the axial direction. This equation has been used to calculate the frequency at which this broadband random (vortex) noise occurs. For the results presented in figure 4, $u = 530$ ft/sec (161.5 m/sec), $d = 0.24$ in. (0.61 cm), and

$$f = \frac{0.2 \times 530 \times 12}{0.24} = 5300 \text{ cps}$$

approximately the frequency shown for the experimental results. The agreement between calculated and measured results was equally as good for all rotor speeds and configurations.

Nature of Aerodynamic Interactions

The reduction of the discrete frequency noise peaks occurring at the blade passage frequency is of chief concern in compressor noise studies. This discrete noise (figs. 3 and 4) is attributed to the fluctuating forces on the blades. It is due in part to

velocity variations in the stream flow (wake velocity defect) and in part to the interaction of the potential flow field of the rotating and stationary surfaces.

Wake defects.- Figure 5 shows schematically the calculated wake width and the defect in velocity profile across the wake as a function of the distance behind the trailing edge of an airfoil. (See ref. 5.)

The illustration of figure 5 indicates that an airfoil (for example, a rotor blade) cutting through the wake of an upstream airfoil (for example, an IGV) experiences a relatively narrow region of reduced velocity when it is close behind the upstream airfoil but that the wake increases in width and the defect in velocity decreases as the distance between the IGV and rotor is increased. In fact far downstream, the rotor will experience essentially no velocity defect. The chief effect of this reduction in velocity on the conditions at the rotor is to cause a change in incidence angle. This change in incidence angle causes a fluctuating load on the rotor blade, which is an important source of noise radiation. The fluctuating load phenomenon strongly influences the magnitude of the interaction noise at the blade passage frequency. (See fig. 4 at 7290 cps.) In addition to the variations in velocity which produce pressure fluctuations on the rotor blade, there is a turbulent wake interaction as one row of blades interacts with the preceding row. Increased spacing has a beneficial effect in reducing the noise from each of these source mechanisms.

For interaction noise due to velocity defects in the wake, the results of reference 1 indicate that substantial reductions in the discrete tones may be realized with IGV—rotor spacings of up to 4 to 6 chords and that larger spacings have only small effects on the noise. For vortex noise (5200 cps in fig. 4) due to turbulent wake interaction, reference 2 shows that small increases (order of 1/2 chord) are sufficient to realize substantial reductions in noise and that further increases in spacing only give further small reductions.

Resultant velocities.- The preceding discussion has considered the IGV—rotor combination and its noise generation phenomena. In a similar manner the wakes from the rotor blades will produce fluctuating pressures on the stator. The theory of reference 3 predicts similar noise pressure radiation fields for the rotor-stator combination as for the IGV—rotor combination, but the theoretical analysis gives only qualitative and not quantitative results. Some insight into the magnitude of the interaction noise for these tests can be obtained from an examination of the velocity diagram through the compressor as shown in figure 6. The average axial velocity through the duct for the several configurations was 190 ft/sec (57.9 m/sec) when the rotor was operating at 8250 rpm. This is the resultant velocity at the IGV. The calculated velocity relative to the rotor (0.9 tip radius) is 517 ft/sec (157.6 m/sec) and the calculated resultant velocity of the stator vane (0.9 tip radius) is 232 ft/sec (70.7 m/sec). Even though the wake behind the

IGV or behind the rotor blade is of the same form for equal distances downstream (fig. 5), the large differences in the relative velocities produced by this configuration over the rotor and stator (517 ft/sec as compared with 232 ft/sec, fig. 6) suggest that the magnitudes of the aerodynamic forces differ greatly. The higher relative velocities and greater fluctuating aerodynamic forces would in turn be expected to produce higher interaction noise levels for the IGV—rotor combination than for the rotor-stator combination.

Noise Radiation Patterns

The data presented in figures 7 to 9 are in the form of noise radiation patterns showing the sound pressure level as a function of azimuth angle. These noise radiation patterns were obtained with a narrow band (approximately 10 percent bandwidth) filter analyzer centered at the fundamental blade passage frequency. Radiation patterns are presented for such configurations as the IGV—rotor, the rotor-stator, and IGV—rotor—stator combinations, with close spacing and increased spacing.

Effects of presence of inlet guide vane.- In figure 7 are plotted noise radiation patterns for the rotor alone and for the 62-IGV—rotor combination at 8250-rpm compressor speed. (See insert sketch at top of fig.) There is a large increase in noise level for the IGV—rotor combination compared with that for the rotor alone. For this configuration and at this rotor speed the interaction noise due to the presence of the IGV is large. These data have been discussed in reference 1 and, when compared with the radiation patterns of the IGV—rotor combination calculated by the method given in reference 3, were shown to be generally in good qualitative agreement.

Effects of presence of stator.- Figure 8 compares the noise radiation patterns of the rotor alone with those of a rotor-stator (62 vanes) combination at 8250-rpm compressor speed. The results show there is very little difference in noise levels for the rotor alone compared with those for the rotor-stator combination; this indicates that the additional interaction noise due to the presence of a downstream stator is small. The radiation patterns for the IGV—rotor combination (fig. 7) and for the rotor-stator combination (fig. 8) are similar as predicted by the theory of reference 3.

Effect of increased guide vane spacing on combinations.- Figure 9 shows noise radiation patterns for the IGV—rotor—stator combination with close and increased spacing for a compressor speed of 8250 rpm. Also shown, for comparison, are the radiation patterns for the IGV—rotor combination with increased spacing. With the increased spacing between the IGV and rotor, the presence of the downstream stator is seen to have an additional small beneficial effect on the noise. Figure 9(a) presents the results for the 62-IGV configuration and figure 9(b) presents the results for the 53-IGV configuration. In each case the spacing between the IGV and rotor was increased

6.25 chords by the use of a spacer ring. No change in spacing was provided between the rotor and stator for the IGV—rotor—stator combination since the results of figure 8 indicate that the interaction noise of the stator on the rotor was already small. Examination of the results of figure 9(a) indicates that although the radiation patterns remained essentially the same, large reductions in noise occurred when increased spacing was used. The noise radiation patterns of figure 9(b) for the 53 IGV and 53 rotor blades are widely different in shape from those in figure 9(a) as predicted by the theory of reference 3 and shown experimentally in reference 1. As for figure 9(a), increased spacing gave large reductions in the noise level with essentially the same lobe pattern. The reductions in noise obtained with IGV—rotor spacing shown in figure 9 are at the fundamental blade passage frequency. For the configurations of the present investigation, equally as large or larger reductions were generally obtained at the first and second harmonics of the blade passage frequency.

The magnitude of the interaction noise is dependent on the resultant velocities over the fixed and moving surfaces, which vary with compressor design, and the results of the present investigation relate to the influence of the inlet guide vane and downstream stator on the noise produced by a single-stage rotor. However, it would be expected that these same noise generating mechanisms exist for a multistage compressor. The work of reference 6 suggests that for a multistage compressor the noise radiated from the inlet results mainly from the first stage, and the contributions from the latter stages are small by comparison. Therefore, substantial noise reductions for multistage axial-flow compressors may be realized with increased spacing between only the IGV and first-stage rotor.

CONCLUDING REMARKS

Noise measurements have been conducted with a single-stage axial-flow compressor in an anechoic chamber. The investigation included studies of the noise associated with the rotor alone, the IGV—rotor combination, the rotor-stator combination, and the IGV—rotor—stator combination with close and increased spacing.

The results indicate that the radiated noise levels due to the interactions of rotor and downstream stator vanes are appreciably less than those due to inlet guide vane and rotor interactions. Noise reductions result from increased IGV—rotor spacing for the IGV—rotor combination and these reductions are further increased with the addition of

the downstream stator. It has been possible to relate variations in the noise levels with a knowledge of local flow conditions in the vicinity of the blades.

Langley Research Center,
National Aeronautics and Space Administration,
Langley Station, Hampton, Va., December 22, 1966,
126-16-03-01-23.

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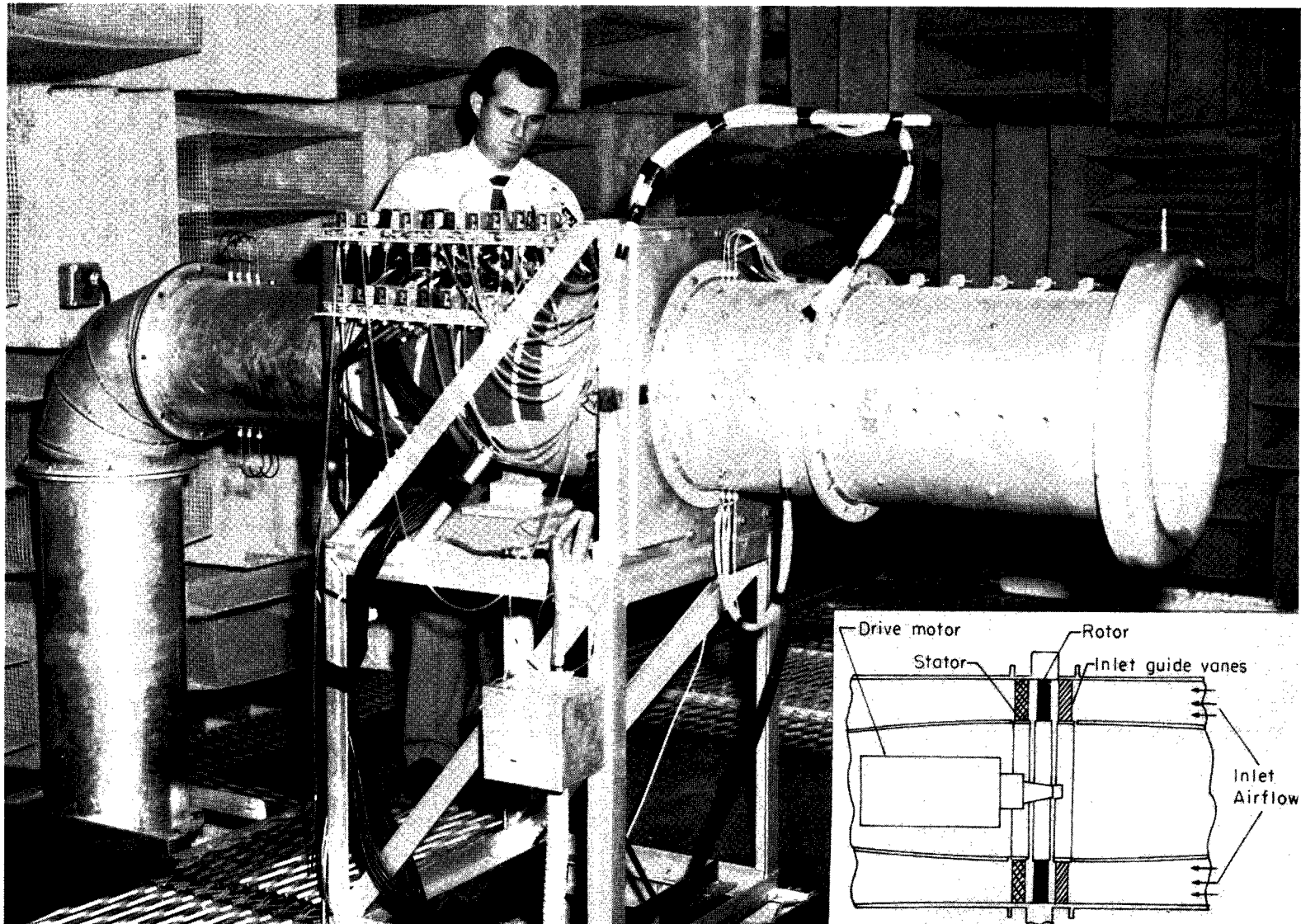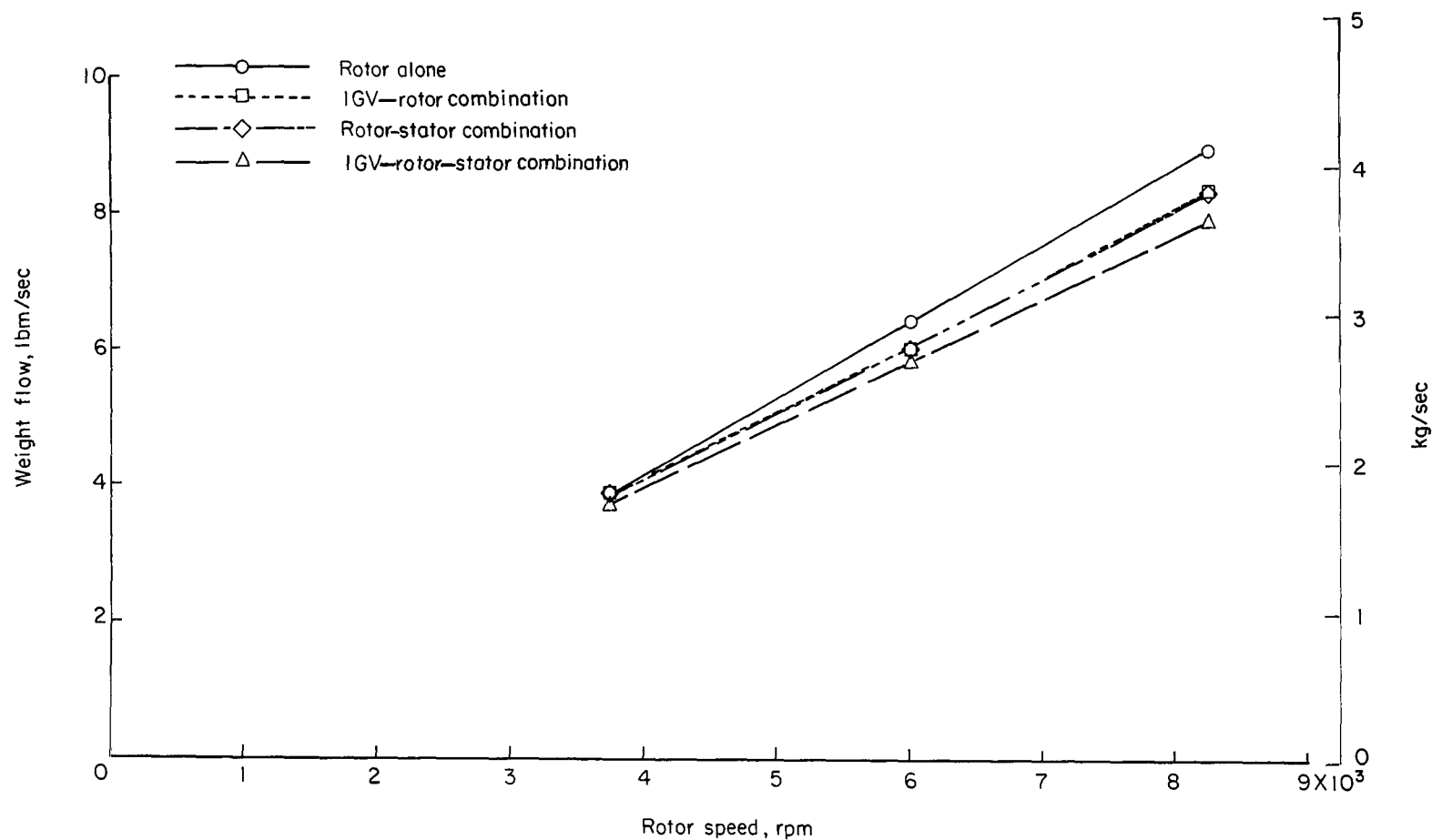
Figure 1.- Test compressor mounted in anechoic cell.

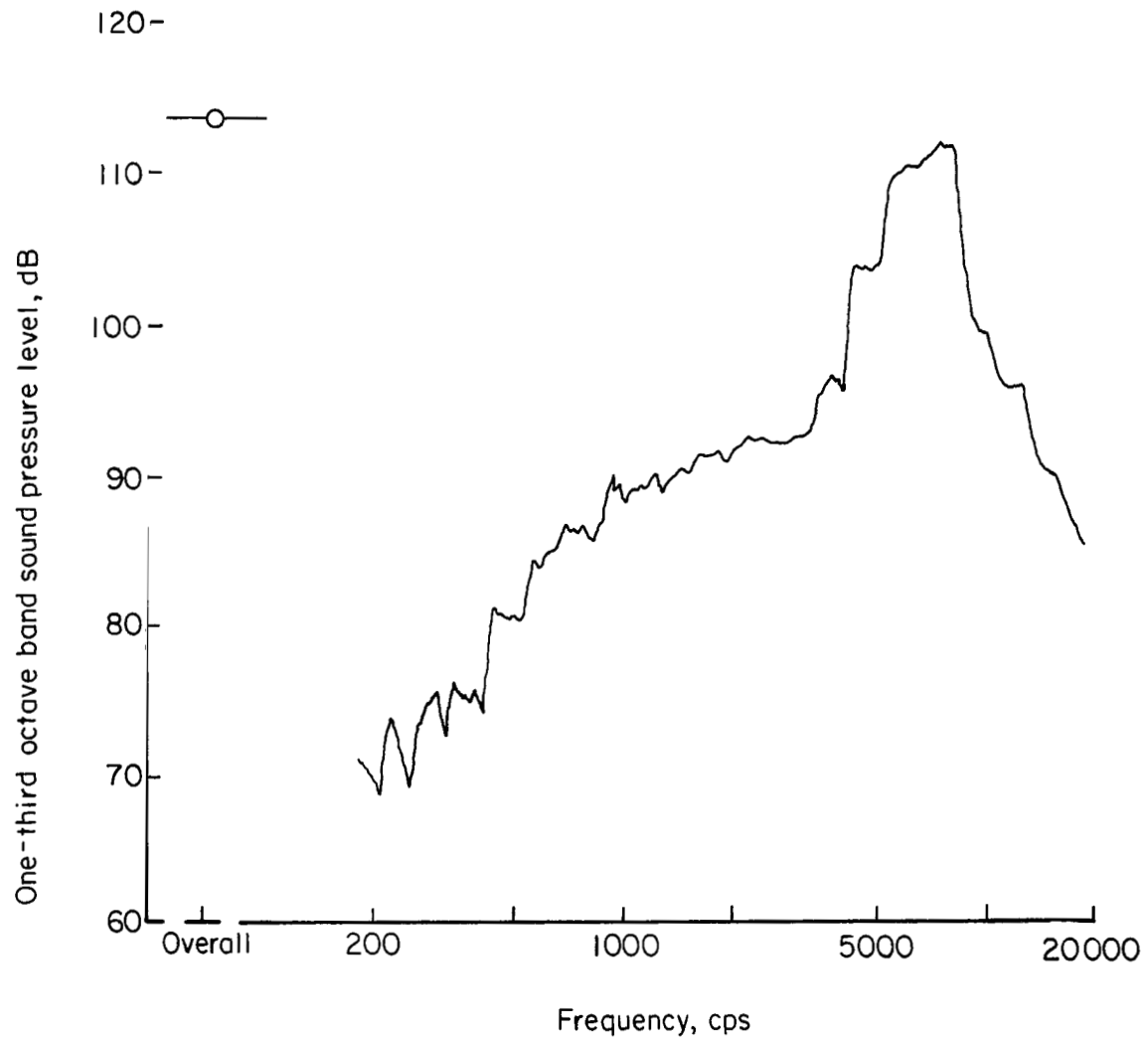Figure 2.- Range of weight flows for all configurations.

Figure 3.- One-third octave band spectrum for the 62-IGV—rotor—stator combination as measured at 30⁰ azimuth and rotor speed of 8250 rpm.
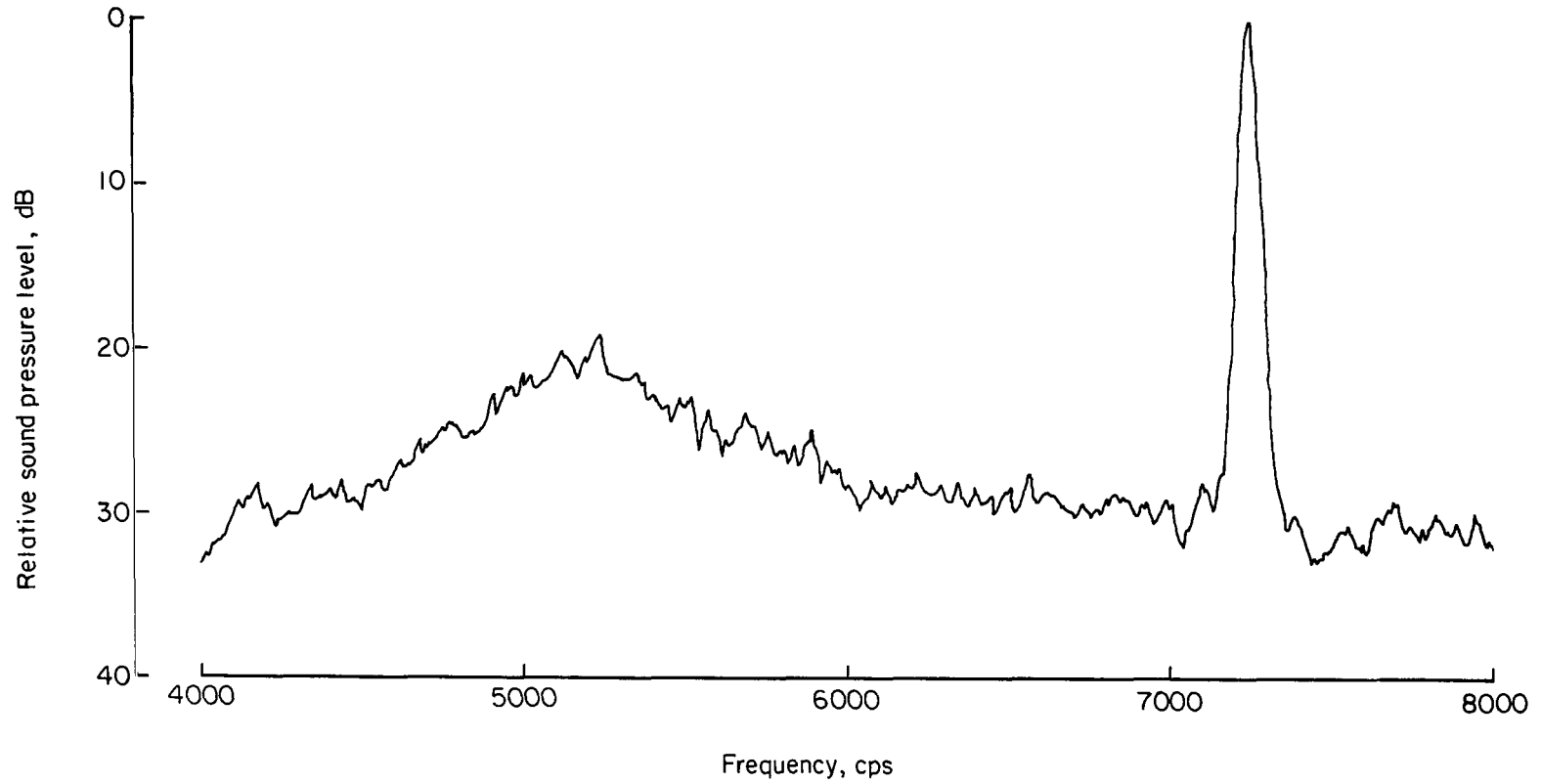
Figure 4.- Frequency analysis, using 50-cps constant band filter, for frequency range containing the fundamental blade passage frequency as measured at 30⁰ azimuth and rotor speed of 8250 rpm.
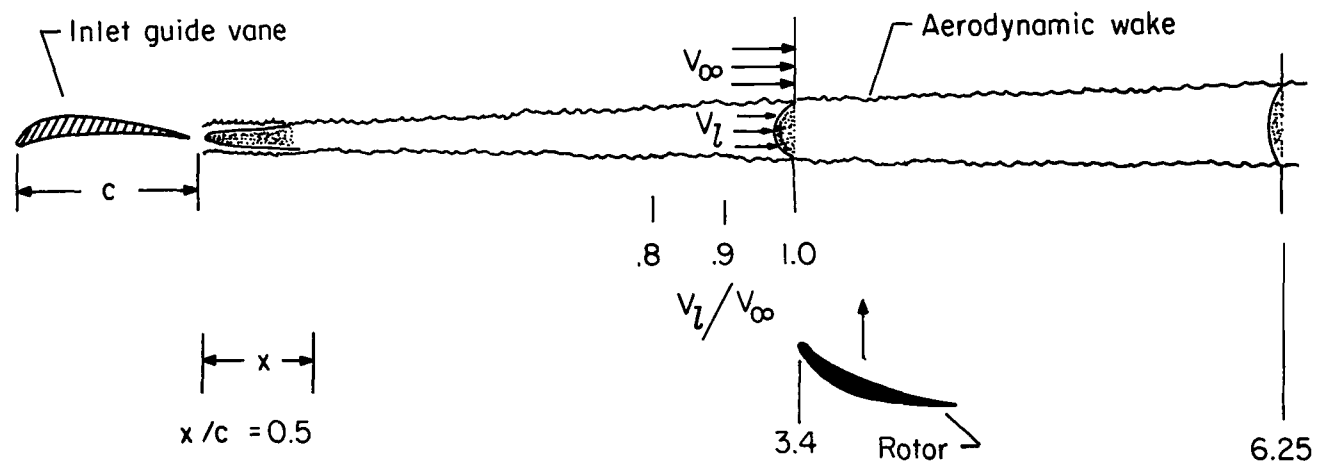
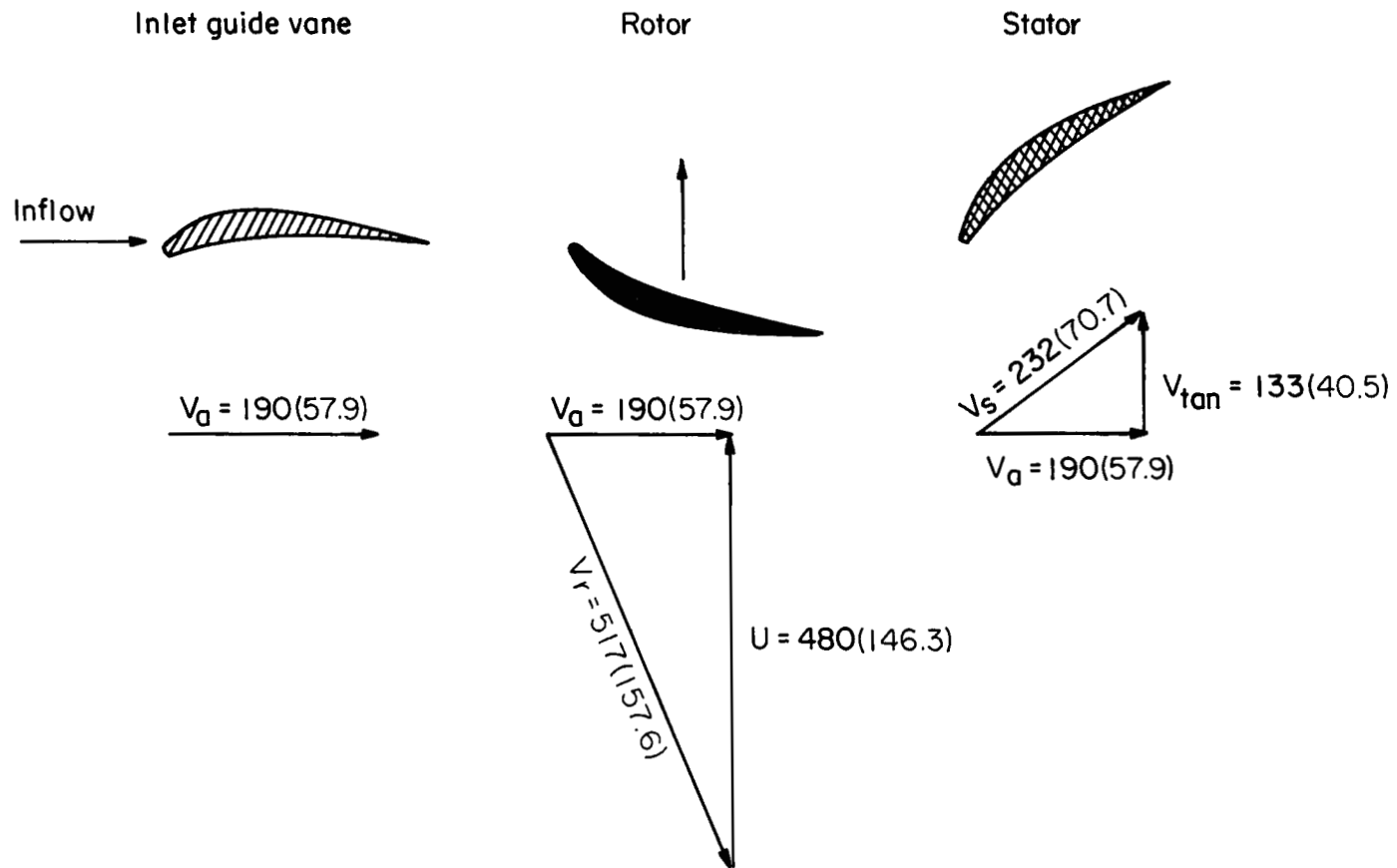Figure 5.- Wake velocity defects at different spacings.

Inlet guide vane   Rotor   Stator

Inflow

$V_a = 190(57.9)$

$V_a = 190(57.9)$

$V_s = 232(70.7)$

$V_{tan} = 133(40.5)$

$V_a = 190(57.9)$

$V_r = 517(157.6)$

$U = 480(146.3)$

Figure 6.- Velocity diagram through compressor at 0.90-in. (2.29 cm) tip radius. Velocities are given in ft/sec (m/sec).

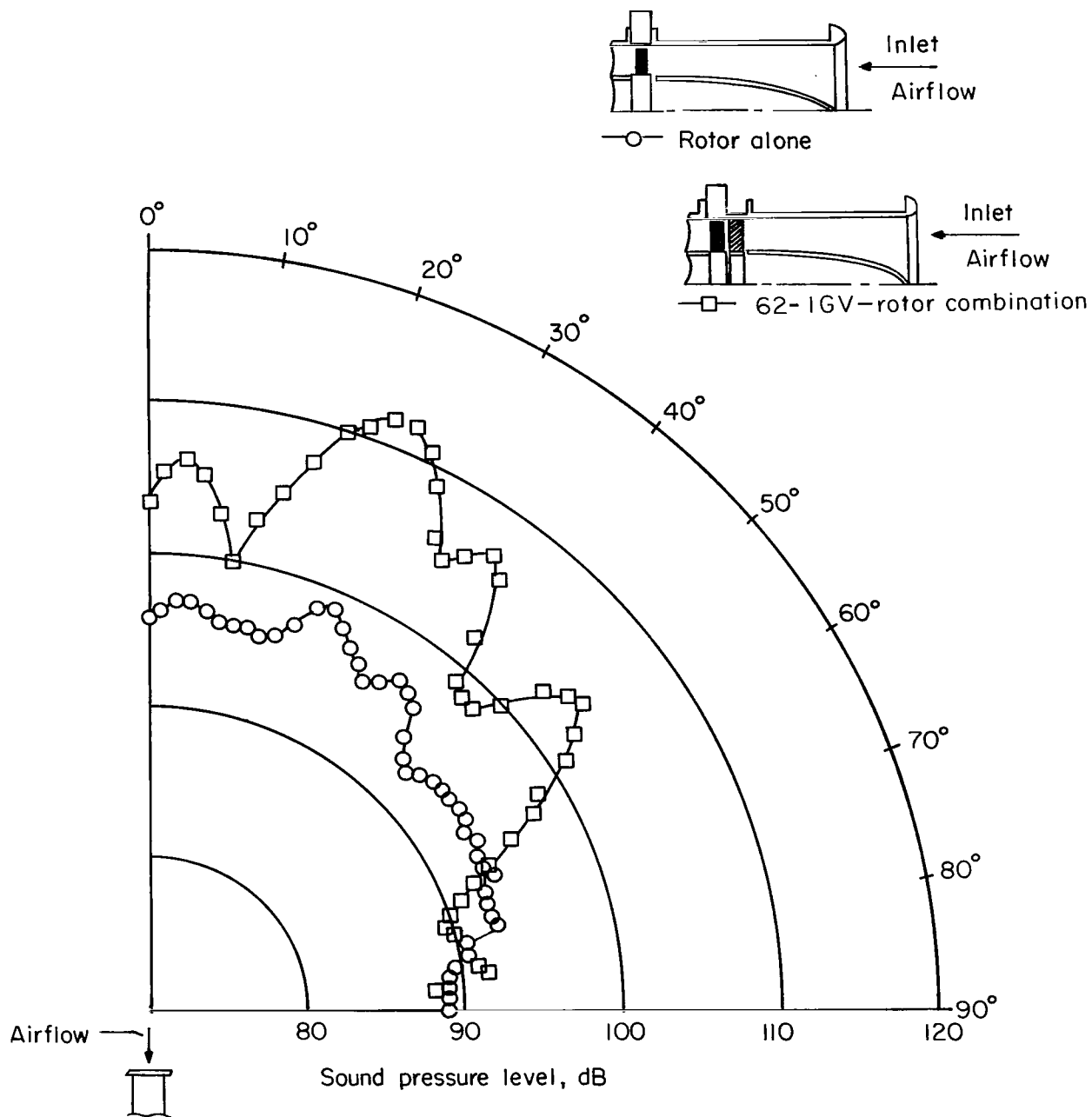Figure 7.- Effect of IGV on noise radiation patterns. Narrow band analysis at fundamental blade passage frequency; compressor speed, 8250 rpm.
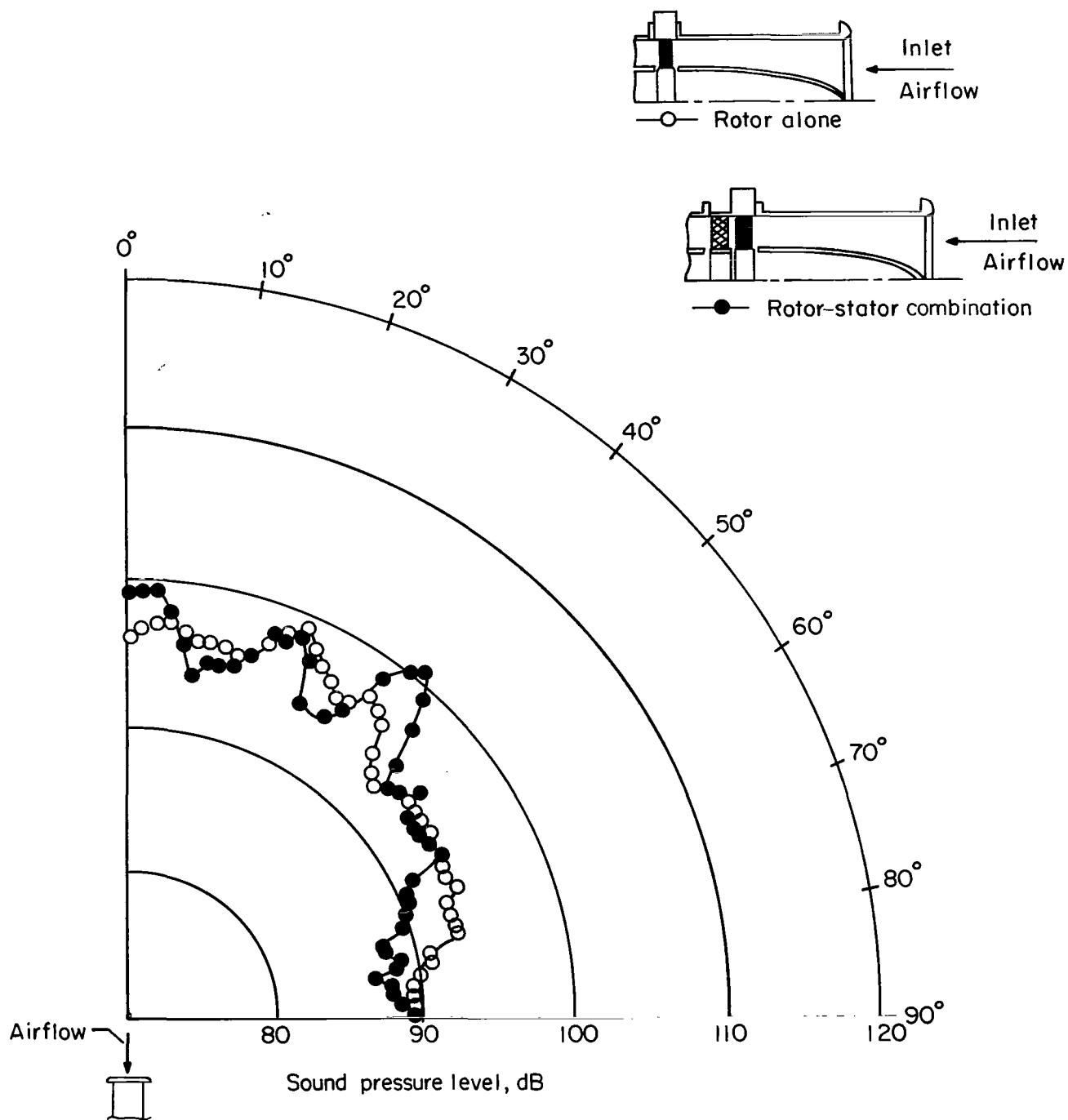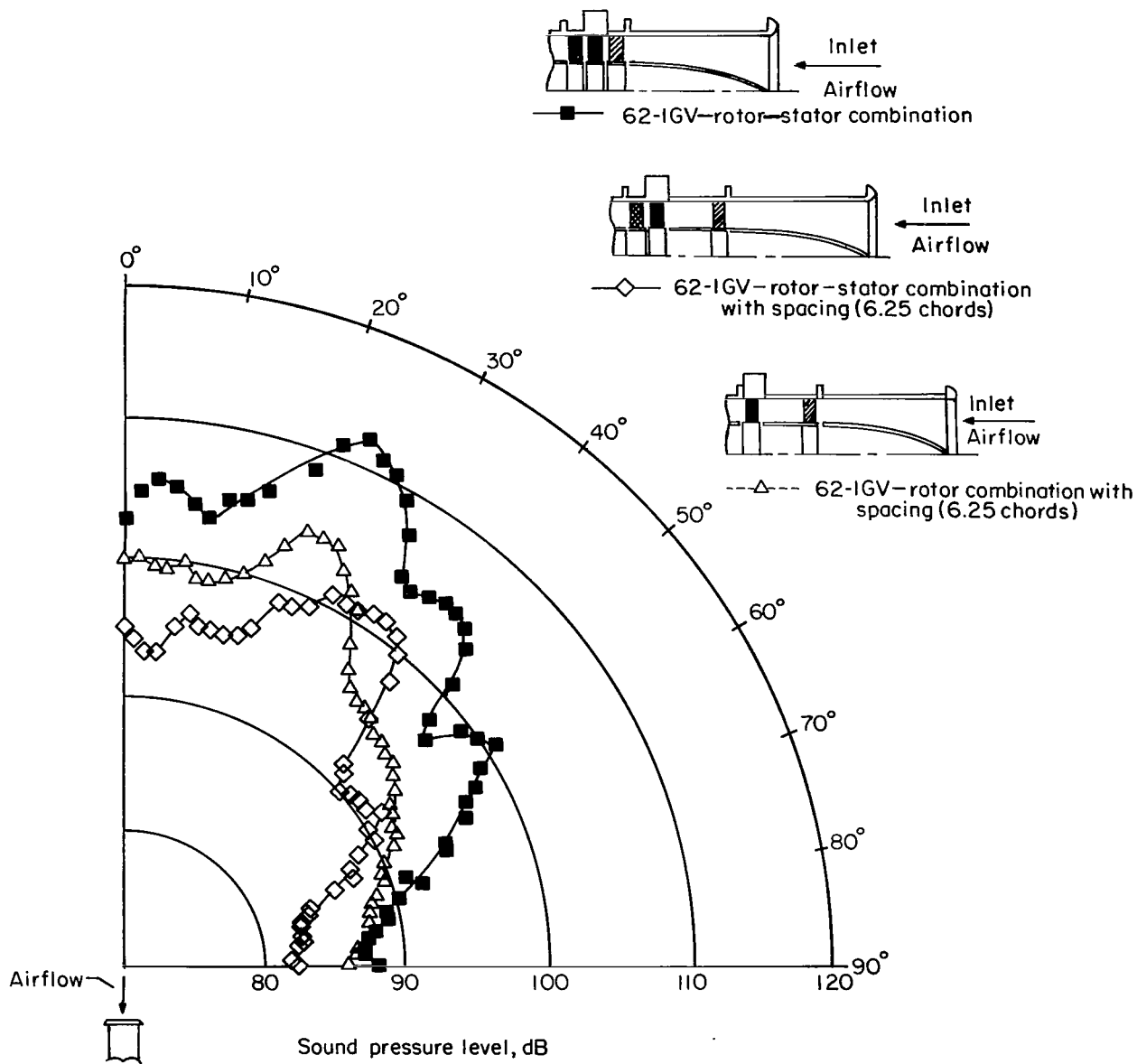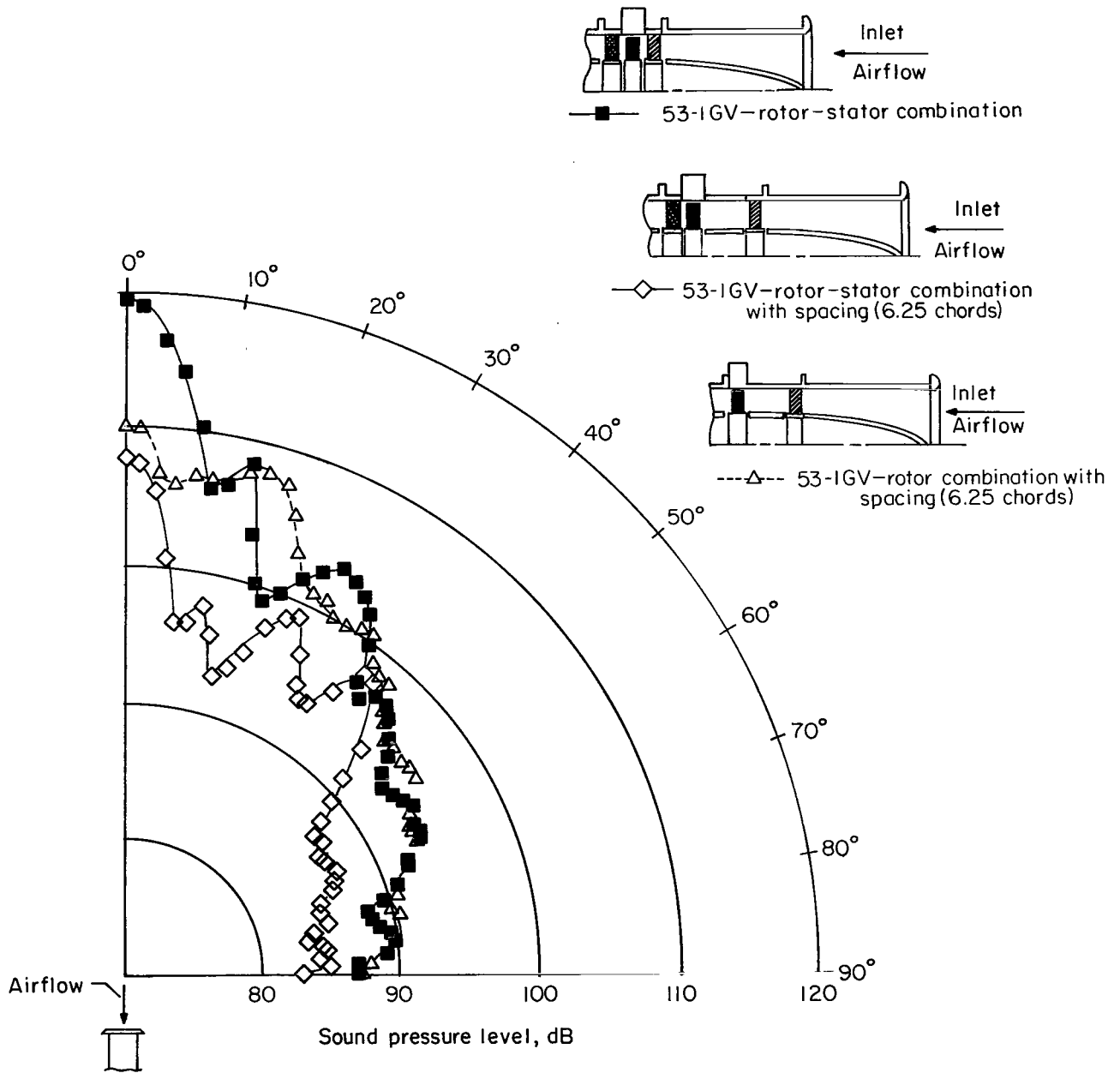
Figure 8.- Effect of stator on noise radiation patterns. Narrow band analysis at fundamental blade passage frequency; compressor speed, 8250 rpm.

(a) 62-IGV configuration.

Figure 9.- Effect of increased IGV—rotor axial spacing on noise radiation patterns. Narrow band analysis at fundamental blade passage frequency; compressor speed, 8250 rpm.

Inlet
Airflow

—■— 53-IGV—rotor—stator combination

Inlet
Airflow

—◇— 53-IGV—rotor—stator combination
with spacing (6.25 chords)

Inlet
Airflow

---△--- 53-IGV—rotor combination with
spacing (6.25 chords)

0°   10°   20°   30°   40°   50°   60°   70°   80°   90°

Airflow

80   90   100   110   120

Sound pressure level, dB

(b) 53-IGV configuration.

Figure 9.- Concluded.

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